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Abstract

Rolling bearings are the second most used machine components. They work in what it is called elastohydrodynamic lubrication regime. The geometry of rolling element bearings makes the direct measurement of the lubricant film thickness a challenging task. Optical interferometry is widely used in laboratory conditions for studying elastohydrodynamic lubrication however it cannot be used directly in rolling element bearings thus the only suitable methods are electrical techniques. Of these, film thickness measurement based on electrical capacitance of the contacts has been used in the past by a number of authors. One of the limitations of the capacitance method, when used in rolling bearings, is that it cannot distinguish between the contacts of every rolling element and raceway on one hand and on the other between the inner and outer ring contacts. In the present study the authors used an original test rig which can measure the film thickness for only one ball and separately for the inner and outer rings of a radial ball bearing. This paper thus shows for the first-time results of the lubricant film thickness, at the inner and outer raceways, in grease lubricated rolling bearings.

Keywords

Elastohydrodynamic lubrication, grease, film thickness, rolling bearings, electrical capacitance

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Introduction

Rolling elements bearings are found in most machines and equipment which include parts in relative rotation (or translation) which makes them the second most numerous machine elements in use nowadays. Their use in applications extending from mining to medical drilling equipment, from space crafts to cars and from toys to wind turbines, makes the precision, reliability, and durability in operation of rolling element bearings vital to our day to day lives. One element of greatest importance for the proper operation of bearings is the lubricant, more precisely the lubricant film formed between the rolling elements and the raceways. This ensures that the metallic surfaces are in no direct contact, avoiding seizure, premature wear, and catastrophic failure of the elements and of the whole assembly they are part of. Understanding the mechanisms of lubricant film formation and monitoring film thickness could prevent such failures which most often lead to failure of complex and expensive equipment or machinery.

The small contact area and large pressures between rolling elements and raceways due to the non– conformity of their surfaces lead to very thin lubricant films, a regime of lubrication known as elastohydrodynamic (EHD). The characteristics of EHD contacts make the direct measurement of the lubricant film thickness in machine elements working in this regime of lubrication difficult. Nevertheless, precise measurement can be carried out on model contacts, in laboratory conditions, using optical interferometry. The principles of optical interferometry and its various ways in which it is applied to the study of elastohydrodynamic contacts has been published extensively during the past fifty years[1–5] and it will not be detailed in this paper.

Due to the geometry of rolling bearings optical interferometry cannot be applied directly to such machine elements, however, there are studies carried out on ball-on-disc apparatuses which, over the years have enriched our understanding of rolling bearings functioning and lubrication in particular. Wedeven and co-workers used optical interferometry to study the starvation in rolling element bearings.[6] They show that the lubrication condition of the bearing is related

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to the length of the meniscus of the lubricant in the inlet of the contact. According to Lugt[7] more than 90 percent of rolling element bearings currently in use are lubricated with grease thus a special attention was given to this topic by the researchers in the field. Rastegar and Winner[8] carried out a study on the traction and film thickness characteristics of grease lubricants in point elastohydrodynamic contacts. Optical interferometry was used for measuring the lubricant film thickness. They concluded that the traction coefficients of greases were lower than that of the corresponding base oils and that this parameter depended on the viscosity of the base oil and the ellipticity of the contact. They also showed that film thickness of grease in fully–flooded contacts was larger than that of the base oil, but fully–flooded conditions were difficult to maintain. Cann et al.[9] combined optical interferometry and infrared spectroscopy to study EHD film formation by grease lubricants. They showed that soap fibres pass through the contact contributing to the EHD film formation. Soap fibres aligned in the inlet of the contact indicating a loss of the viscosity.

Cann and Lubrecht[10] analysed the mechanisms of rolling element bearings lubricated by grease based on results obtained in simulated, ball on flat contacts. They linked the film formation in the EHD contact to the efficiency of lubricant supply. They conclude that the predominant mechanism for lubricant supply was the reflow of base oil, from grease reservoirs, due to capillary/surface forces. Eriksson and co-workers[11,12] studied the passage of soap particles using optical interferometry and high–speed imaging. They pointed out that soap particles do pass through the contact, even though the number of such particles passing was not related to the soap concentration, proving that the bulk grease does not pass through the contact. They also showed that the soap fibres kept their shape and size during the passage.

Apart from optical interferometry, electrical methods have also been used to study the formation and behaviour of EHD films. Electrical capacitance especially, has been extensively used to evaluate film thickness in EHD contacts, both in model contacts and in real machine elements. The choice of electrical capacitance is natural because this quantity is linked directly to the separation between the solid surfaces in contact, which play the role of armatures of a capacitor. The formula of the capacitance of a plane parallel capacitor has been widely used to extract lubricant film thickness, from capacitance measurements in EHD contacts.

$$C = \varepsilon_0 \varepsilon_r \frac{A}{h} \qquad (1)$$

In this relationship ε_0 is the absolute dielectric permittivity or the permittivity of vacuum, ε_r is the relative dielectric permittivity, or the dielectric constant of the dielectric material (in this case the lubricant inside the contact), A is the surface area of the capacitor and the h the separation between the solid surfaces. The use of this formula for EHD contacts was justified by the fact that, at low films and/or large loads, the lubricant film is almost parallel over the most part of the contact area, fact predicted theoretically[13,14] and proven by optical interferometry studies mentioned above.

Electrical capacitance has also been used to evaluate film thickness in internal combustion engines' piston–ring contacts,[15–17] machine components such as gears and cams[18–19] or thrust bearings,[20] but this paper will only focus on its application to study EHD lubrication in general and that of rolling element bearings in particular. In one of the first studies of this kind Crook[21,22] carries out a comprehensive analysis of elastohydrodynamic lubrication based on the capacitance measurement of the line contacts of rollers test rigs. Later he used sub–surface transducers and electrical capacitance to evaluate, lubricant film shape and thickness of a line contact.[23] Sub–surface transducers have been later used to study not only lubricant film thickness and shape but also pressure distribution and temperature rise in the contacts.[24–30]

Dyson and co–workers also used a disc machine and electrical capacitance to study the film thickness in line contacts for various lubricants and working conditions.[31] They found a close correlation between their experimental results and the theoretical predictions by Dowson and Higginson formula for EHD line contacts film thickness.[32] In addition, they made a detailed analysis of the capacitance of the conjunction between the rollers, distinguishing three regions which contribute differently to the total capacitance: the EHD contact itself, the inlet and the outlet regions. This approach has since been used in most of the studies which employed electrical capacitance to investigate EHD films. Dyson and Wilson extended this research to the study of lubrication by grease of the contact between two rollers.[33] They found that the grease gives thicker film thickness at the beginning of the test but thinner films as the test progresses and attribute this behaviour to the viscoelastic properties of the lubricants.

Jablonka et al.[34] carried out simultaneous measurements of film thickness, by optical interferometry and capacitance measurements in an oil–lubricated contact between a chromium–coated glass disc and a steel ball. This technique allowed the calibration of the capacitance technique, which was later used for the study of film formation between steel elements.

Lubricant film measurement in rolling bearings

Published measurements of lubricant film thickness, carried out directly on rolling element bearings are relatively rare. Wilson uses the methodology developed in[33] to investigate comparatively the film thickness in rolling bearings lubricated by grease and by their base oils.[35] He treated the bearing under test as an electric circuit formed by a number of capacitors in series and in parallel. This is illustrated in Figure 1, which is adapted from literature.[35] The capacitances of the balls are connected in parallel; each ball has two capacitors (one for the inner ring contact and one for the outer ring contact) connected in series; each contact between one ball and one of the raceways is treated as three capacitors in parallel–the capacitance of the Hertzian contact, the capacitance of the inlet zone and that of the outlet zone. The dielectric constant of the lubricant inside the contact is adjusted to account for the larger pressure while the dielectric constant of the outlet region is estimated by considering that the lubricant film splits in two halves which stay attached to the solid surfaces, and the gap in between is filled with air.

Heemskerk et al.[36] developed a novel technique and device to study, what they called "measurement of lubricant condition". With this technique they evaluated the metallic asperity contacts and the "lift– off" speed when the two metallic surfaces become completely separated. They consider the lubricant film thickness as a function of a constant "c", which takes different values for inner and outer rings contacts, and for different types of bearings. The same instrument, but an approach similar to Wilson's[35] was adopted by Leevers and Houpert[37] who measured the film thickness in deep–groove and spherical bearings. They adjust the film thickness calculated from capacitance measurement, by a thermal correction factor. One of their main conclusions was that a film which separates completely the metallic surfaces was formed, even in marginal lubrication condition. They hint, as a possible reason of this result, at the concept of functional filtering which, applied to their case, would lead to smaller asperities and larger slopes which would then explain the results obtained.

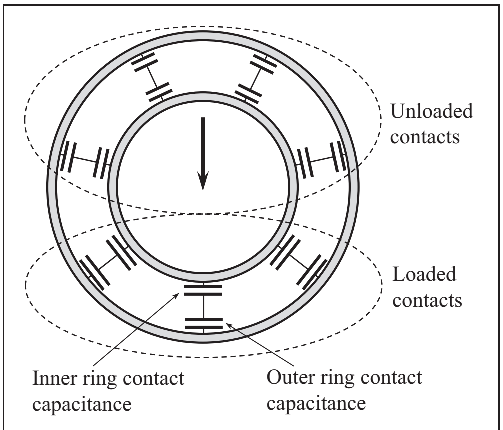

Figure 1. Capacitance of a rolling element bearing contacts.

More recently experimental findings based on optical interferometry technique from Morales-Espejel et al.[38] Cen et al.[39] and Concalves et al.,[40] have shown that grease can provide significantly thicker film comparing with its corresponding base oil at rather low speed under fully flooded condition; in this case the film thickness is dominated by the corresponding base oil viscosity at high speed region. This is attributed to the high effective viscosity of grease at lower rolling entrainment speed, which has been further physically explained in detail by Kochi et al.[41] with the discussion on grease preferential at different phases.

Steady state grease film forming and flow behavior under both fully flooded and starved condition have also been showed by the experimental studies carried out by Chen et al.[42] and Vengudusamy.[43] At the beginning of 2020, the authors of the current paper published a paper focusing on the effect of normal sinusoidally variable loading on grease lubricated EHD contact;[44] it was experimentally found that vibrations play a positive role in track replenishment of a grease lubricated circular contact.

Wittek and co–workers employed electrical capacitance to measure film thickness in a deep – groove bearing loaded axially, in order to ensure uniform conditions in each contact.[45] They make the simplifying assumption that the film thickness at the inner and outer rings contacts are equal. They also use the finding of Bartz[30] according to whom the total capacitance of the contact is 3.5 times larger than Hertzian contact capacitance. Capacitance measurement in rolling bearings, in an experimental arrangement very similar to the one in[45] was employed by Baly et al.[46] in a study which compared film thickness and friction in rolling bearings to those obtained in model contacts using optical interferometry. They conclude that the two sets of tests are complementary and are both needed in order to obtain a full picture of the lubrication mechanisms in bearings used in industrial applications.

In a recent study Cen and Lugt[47] measured grease lubricant film thickness in a sealed deep groove ball bearing using an electrical capacitance method-based rig under variation of load and speed conditions. Experimental working parameters showed close relationships with industrial applications, especially the prolonged testing durations which allowed the film thickness recovery to be clearly detected under starvation condition.

An overview of the capacitance tests in rolling element bearings unveils a number of assumptions and simplifications usually made in extracting lubricant film thickness from capacitance measurements. These are:

- The capacitor formed between rolling element and raceway is considered a parallel plate capacitor

- The film thickness is equal at the inner and outer ring contacts
- The capacitances of the rolling elements are connected in parallel
- The capacitance of one contact is formed by the capacitance of the Hertzian area, the inlet and side regions and the outlet region, connected in parallel
- The Hertzian contact capacitance is a proportion of the total measured capacitance
- The dielectric constant of the fluid inside the Hertzian area is adjusted for pressure
- The dielectric constant of the fluid in the outlet region is either taken as unity, for air, or is considered a mixture of air and oil
- The region filled with lubricant covers the projected area of the ball

The calculation of the film thickness from the total measured capacitance is complicated by the fact that the load varies with the position of the ball, thus the Hertzian contact area is different not only between the inner and outer ring contacts, but between various balls as well. Another uncertainty is the exact value of the internal clearance of the bearing. This is usually given as a range of values but differs from the operation clearance due to the fit used during assembling and variation of temperature. The value of the clearance chosen strongly influences the load distribution between balls and thus the contact area. Some of these uncertainties have been addressed by Jablonka et al.,[48] for example the one regarding the ratio between the total and Hertzian contact capacitance, based on simultaneous capacitance and optical film thickness measurement.

According to Lugt[7] grease lubrication is still not yet well understood and as shown previously the great majority of rolling bearings are lubricated by grease thus in this paper the authors use an original test rig which is able to discern between the capacitance of the inner and outer ring contacts, focusing on a comparison of the film thickness measurement between three greases and their base oils.

Experimental rig and materials

The bearing tested is a deep–groove ball bearing 6306ETN9 fitted to a rotating shaft, which receives motion from an electric motor. The cage of this bearing is made of an insulating, thermoplastic material (glass fibre reinforced polyamide PA66) thus it does not interfere with the measuring electrical circuit. All the rolling elements, except the test ball were replaced with electrically insulating balls made of silicon nitride, of the same diameter. The steel test ball was positioned at the maximum load point and held stationary by fixing the cage. As the cage was fixed, in order to allow the bearing to continue to operate properly the outer ring of the bearing was made to rotate, by fitting it into a larger bearing, which, in its turn has the outer ring fitted to a frame. This frame is part of the loading system, which also comprises a lever and weights.

A schematic of the main part of the test rig is shown in Figure 2. The test ball has a hole through its middle and a shaft fitted to it. This shaft is attached to a mercury slip–ring for connecting the ball into the electric circuit. The inner and outer bearing rings are electrically insulated from each other and from the rest of the rig. The bearing rings are connected to the electric circuit via suitably placed slip–rings. In this way the contacts of the ball with inner or outer rings are separated in the electric circuit and their capacitance, and subsequently the film thickness, can be measured independently.

In this arrangement the force loading the contacts under study can be evaluated with good confidence because the ball is fixed. In order to evaluate this force a numerical procedure was devised. As the elastic constant of the test ball is different from that of the other balls of the test bearing, which are made out of silicon nitride, the load carried by this ball is smaller than that corresponding to a standard bearing where all balls are made out of steel. The latter was calculated using the formulas derived by Hamrock and Anderson.[49] Figure 3 shows the difference of load carried by the test ball for a standard bearing and for the modified bearing.

The capacitance of the contacts was measured by an impedance phase–shift analyser Solartron 1260. The frequency of the current used was 100 kHz and the voltage 50 mV. The voltage was set to this low value to avoid current breakthrough the lubricant film, as this is expected to be thin in grease lubricated bearings, where starvation is likely to occur.

The load on the bearing used in the tests was 2000 N, thus the load carried by the test ball was calculated at 1108 N. The maximum Hertzian pressure at the inner ring contact is 1.65 GPa and that at the outer ring 1.44 GPa. The tests were carried out at

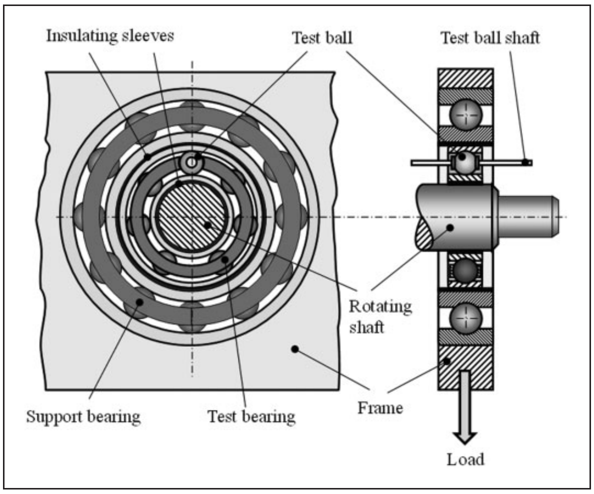

Figure 2. Schematic of the bearing arrangement.

ambient temperature, but the temperature variation was monitored by thermocouples placed under the rings corresponding to the location of the ball. Figure 4 shows the variation of the temperature of the inner and outer rings for a sweep of speed.

Three greases and their base oils were tested for a range of speed of the shaft between 150 rpm and 937.5 rpm. The rotational speed of the test bearing shaft, that is that of the inner ring, was measured by a non-contact digital laser tachometer, which allowed the accurate control of angular velocity of the bearing. Table 1 shows some of the characteristics of the tested lubricants. The reasons for choosing these greases are because their base oils have different viscosities and they also have different composition, although the study of the effect of greases' composition upon film thickness was not the objective of this paper.

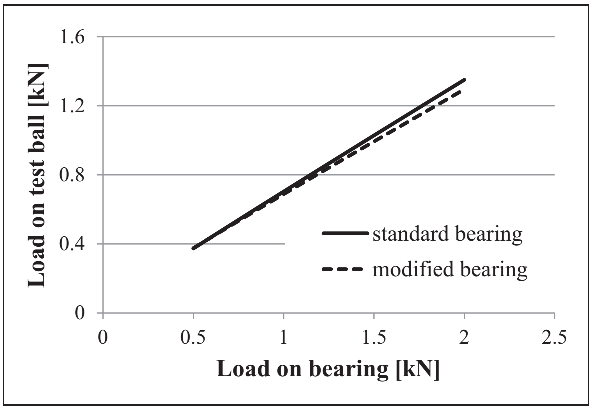

Figure 3. Load carried by the test ball vs total load on bearing.

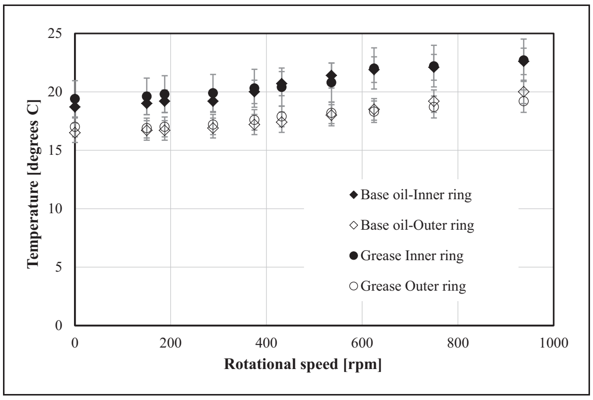

Figure 4. Variation of temperature during test.

The film thickness evaluation procedure was as follows. The test bearing was thoroughly cleaned with toluene and iso-propyl alcohol, allowed to dry and fitted onto the rig. A small quantity of oil, 5 drops from a pipette, was initially poured onto the raceways of the bearing and the shaft was rotated few turns by hand, without load, in order to spread the oil over the whole surface of the raceways. The electric motor was switched on and the load was slowly applied. The capacitance of the inner and outer contact was measured and recorded in turn. Ten readings were averaged for each measurement and the tests were repeated three times. Few drops of oils were added with a pipette, the speed of the shaft increased by a step and the procedure was repeated. The base oils were tested in the first step, followed by the corresponding grease.

Results and discussion

In normal operation of rolling element bearings, where the outer ring is usually fixed, the entrainment velocity at both contacts is given by the relationship:

$$U = \frac{1}{2}\omega_i R\left(1 + \frac{r}{R+r}\right) \tag{1}$$

In this equation ω_i is the angular velocity of the shaft, R and r are the radius of the inner raceway and rolling element (ball in this case) respectively. In the present rig the cage is fixed and both rings are rotating. Assuming that the there is no sliding between the loaded balls and the raceways, the surface speed of each of these elements (balls, inner and outer raceways), are equal. This is a reasonable assumption as the EHD lubricant film, at the contacts between the balls and raceways will experience just enough shearing to allow it to develop shear stresses and thus tangential forces which will make the balls and subsequently the outer raceway rotate. This is seen in Figure 5.

Because the axis of rotation of the ball is fixed, the surface velocity of the elements is $\omega_i R$ thus the

Table 1. Properties of tested lubricants.

Grease	Base oil	Base oil viscosity [mm2/s]	Thickener (weight percentage)	Additives	NLGI Group
Grease 1	Synthetic ester oil	22.8 @ 40°C 4.8 @ 100°C	LiSt (5%–15%)	Barium compound (5%)	3
Grease 2	Synthetic hydrocarbon oil	41.5 @ 40°C 7.4 @ 100°C	Urea (10%–20%)	Zinc compound and barium compound (10%)	3
Grease 3	PAD1450	80.4 @ 40°C 12.3 @ 100°C	Diurea -	-	2

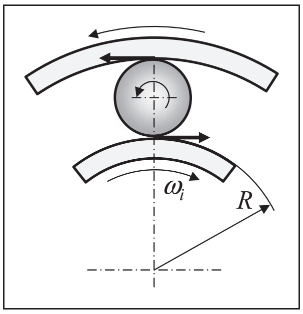

Figure 5. The kinematics of the EHD contact elements.

entrainment velocity, at each contact, is given by:

$$U = \omega_i R \tag{2}$$

As seen the entrainment velocity is larger in the current arrangement, but because this velocity is the same for both inner and outer contacts, the findings of this study will still apply to normal operation of rolling bearings. It is also obvious that there is no need to measure the rotational velocity of the outer ring and only the inner ring velocity is needed. This was measured as explained above.

Figure 6 shows the capacitance variation with the angular velocity of the shaft, for the base oil of the three greases tested.

As expected the capacitance decreases with increasing the entrainment speed, thus with increasing the film thickness. At the same time, the capacitance of the inner ring contact is larger than the outer ring contact, indicating thinner film at the former contact. The difference between the two contacts is not the same for all the oils; the base oil of grease 3, which has the highest viscosity, showing a relatively little variation of this difference, with entrainment speed, between about 35 and 50 percent. The least viscous fluid, (the ester oil of grease 1) shows a very large variation of the difference between the inner and outer ring; at lower speeds, the inner ring contact capacitance is about 2.6 times larger while at larger speeds about 1.5 times larger. For the synthetic hydrocarbon oil of grease 2 this difference is about 40 percent at lowest speeds to less than 10 percent at largest speed.

From the raw capacitance data, the film thickness for the inner and outer rings contacts was extracted, following a procedure detailed in literature.[40] In brief, this procedure involves the evaluation of the capacitance contribution of the region surrounding the Hertzian contact first. This region is divided in three parts: inlet, outlet and sides as shown in Figure 7 as an example for the ball/outer ring contact. This figure is an adaptation from.[48] The inlet region is considered fully flooded with oil up to a distance where the gap between the solids is nine times the central film thickness of the contact. This distance is suggested by Wedeven and Cameron[6] as the minimum distance

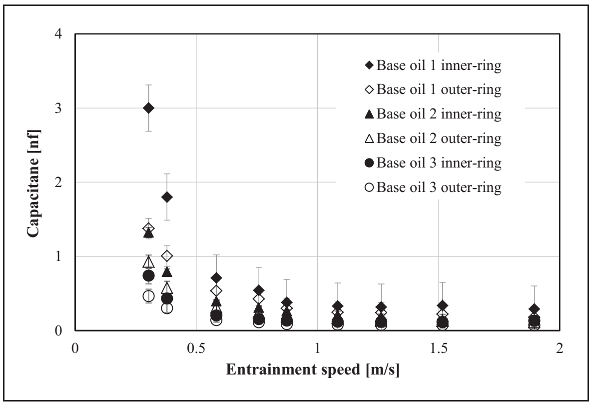

Figure 6. Capacitance of inner–outer rings, for base oils.

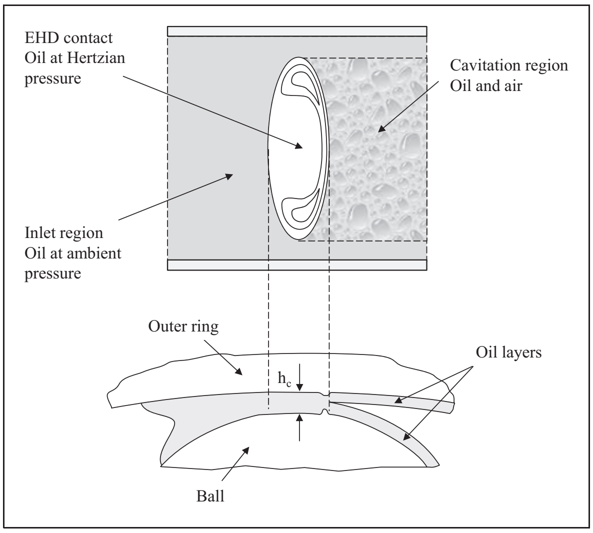

Figure 7. Schematic representation of the contact for capacitance evaluation.

which ensures fully flooded condition. The side regions cover all the distance from the contact edge to the rings' shoulder. Finally in the outlet region, where cavitation occurs, it is considered that the lubricant film is split in two equal parts that adhere to the solid surfaces and the gap in between is filled with air. The capacitances of these regions and that of the Hertzian contact are connected in parallel, thus the total measured capacitance minus the stray capacitance is their sum.

$$C_t - C_{str} = C_{in} + C_{out} + C_H \tag{3}$$

The gap between the solid surfaces, inside the contact, is considered constant, an approximation accepted by all researchers using this method. For thin films, which is the case in these experiments, optical interferometry measurements have indeed shown this to be a valid approximation.[3–6]

Capacitance data shown in Figure 6 were then used to extract lubricant film thickness for the inner and outer rings respectively. Figure 8 shows the measured

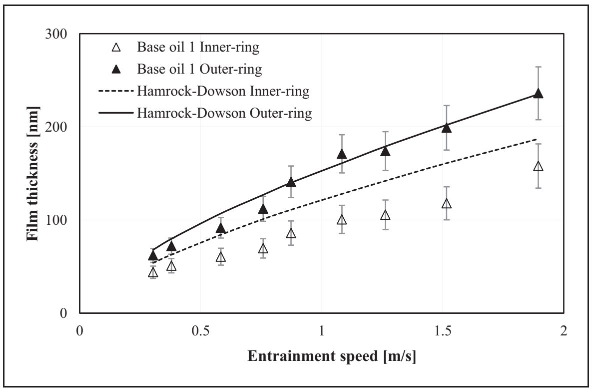

Figure 8. Film thickness vs entrainment speed for base oil of grease 1.

Table 2. Dielectric constant of tested lubricants at 25 °C and low frequency.

Lubricant	Base oil	Grease
Grease 1	2.92	3.66
Grease 2	1.92	1.98
Grease 3	1.96	2.06

film thickness for the base oil of grease 1. The error bars indicate the spreading of the film thickness based on three repeated measurements. For reference, the film thickness results estimated by Hamrock & Dowson formula[50] are also shown. The viscosity of the lubricant in this formula was adjusted, at each speed, to account for the small increase of the temperature over the duration of the test. The dielectric constant of the oils and greases was taken from Nagata[51] as showing in Table 2.

As seen the film thickness extracted from capacitance measurement is smaller than the theoretical one. The same trend was found for all the base oils tested. The smaller values of the measured film thickness at larger entrainment speed can be attributed to starvation, given the fact that only few drops of oil were used to lubricate the bearing.

The difference increases with speed for the more viscous oil of grease 3, as shown in Figure 9, which supports this explanation. All oils also show smaller film thickness at lower speeds which can be explained by the fact that at thin films, the method of extracting film thickness from capacitance is very sensitive to the exact dimensions of the contact and to the contribution to the total capacitance of the region outside of the contact.

The main aim of this research was to evaluate the ratio of the lubricant film thickness formed at the inner and outer rings. The graphs in Figures 8 and 9 show a clear difference between inner and out rings film thickness, for the entire range of speeds employed in these tests. At lower speeds, the difference between outer and inner film thickness is on average about 38% of the inner ring thickness. At higher speeds,

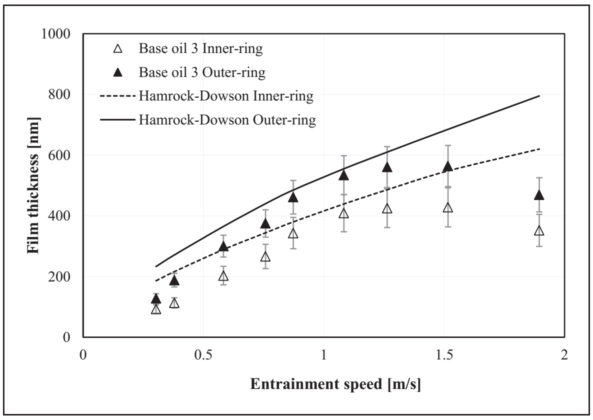

Figure 9. Film thickness vs entrainment speed for base oil of grease 3.

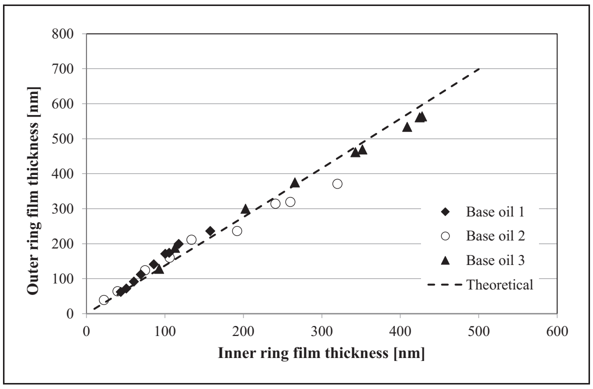

Figure 10. Outer ring film thickness vs inner ring film thickness for three oils.

this difference decreases and becomes smaller than the theoretical one at the highest speed tested. This again can be explained by a combination of shear thinning and the setting of starvation, which makes the inlet boundary move closer to the contact thus cancelling out the effect of the more favourable geometry of the outer ring contact. To better show the ratio between the film thickness at the inner and outer rings contact, the latter is represented as function of the former in Figure 10.

The theoretical line shown is from the ratio of the film thicknesses at outer and inner rings given by Hamrock & Dowson formula. Assuming that the lubricant is at the temperature of the two rings shown in Figure 4, then the theoretical film thickness ratio is given by:

$$\left[\frac{R_o\,(R_i+R_b)}{R_i\,(R_o-R_b)}\right]^{0.476}\left(\frac{\eta_{T_o}}{\eta_{T_i}}\right)^{0.67} \tag{4}$$

where R_b is the radius of the ball, R_i and R_o are the radii of the inner and outer rings in the rolling direction and η_{Ti} and η_{To} are the viscosities of the oil at the inner and outer rings temperature respectively. As it can be seen the general trend is similar, while the slope of the theoretical line is in good agreement with the measured values.

The EHD film formed by grease lubricants was tested with the grease packed all around the bearing, although the bearing had no seals. The test bearing was manually packed based on the case similar to some automotive wheel bearings in real applications. Grease was forced into the spaces between rolling elements with a small laboratory grease packer. The film thickness of grease 1 and its base oil is shown in Figures 11. To note that the film thickness of the grease was calculated from capacitance, using the dielectric constant shown in Table 2.

The first observation is that the film thickness of the grease is larger than that of the base oil for the whole range of entrainment speed. The second is that no marked decrease of the grease film thickness with speed is seen, although the gradient of the curves clearly stops increasing for entrainment speeds over about 1 m/s.

At speeds of approximately between 380 rpm and 480 rpm vibrations of the system were noticed. This is somehow of no surprise because the shaft was relatively long and the test bearing, cage and the loading weights acted as a combined, free mass, which would cause unavoidable bending deflections. We are thus in the typical case of vibrations due to unbalanced rotating masses which will show larger amplitudes around the natural frequency of the system. It is believed that the good supply of oil from the grease reservoir and relatively strong vibrations[44] helped replenish the rolling track thus ensuring full film, or near full film conditions at all speeds. The study of the effect of vibrations on grease lubricant film thickness in rolling element bearings, using the electrical capacitance method in the current system, is a direction of future research which the authors have considered, however this is totally out of the scope of the current study.

It is also worth noting that the grease was not worked before capacitance was measured mainly because, as stated previously the aim of this paper was to evaluate the difference between the film thickness formed at the inner and outer rings of the bearing thus proving the validity of the method.

The relation between the outer and inner rings film thicknesses for the three greases can be seen in Figure 12, together with the theoretical curve for base oil 1, obtained as explained above.

This diagram shows a wider spread of the results in comparison to the base oils, which would be expected as it is known that fresh, un-worked greases often give erratic film thickness values.[41,52] The general trend is similar though. It can also be seen that grease 1 and 3 give larger ratios of the outer to inner contacts film thickness at lower entrainment speeds. As the entrainment speed increase the ratio decreases below the theoretical values which was also observed for base oil, although on a much lesser extent. This trend can be attributed to a number of factors including the fresh grease, vibrations of the test rig and possibly passage through the contact or adherence of thickener to the raceways with effect upon the measured capacitance. A question can be asked whether centrifugal forces, acting on the oil particles can push it against the outer raceway thus the larger film thickness at this contact. Firstly, it should be noted that grease results showed no evidence of starvation, thus on the tracks, there was enough lubricant to ensure fully flooded regime, in the conditions of these tests. Secondly it should be noted that in these experiments the cage of the test bearing does not rotate thus the grease is somehow stationary, not like in real-life rolling bearings where the cage rotates and the grease packed around rotates with it. This means that only the oil film spread on the inner raceway will be subjected to centrifugal forces which would tend to separate it from the solid surface. In this case a very simplistic calculation can be done; assuming that a 2 micrometres thick oil film is spread over the inner raceway, the centrifugal force acting on an element of fluid, one square millimetre, when the ring rotates at the largest speed in these tests of 937 rpm is 0.3 micro Newtons. At the same time, if the surface tension of the oil is taken as 30 mN/m the force holding this oil volume onto the surface due to surface tension is 120 micro

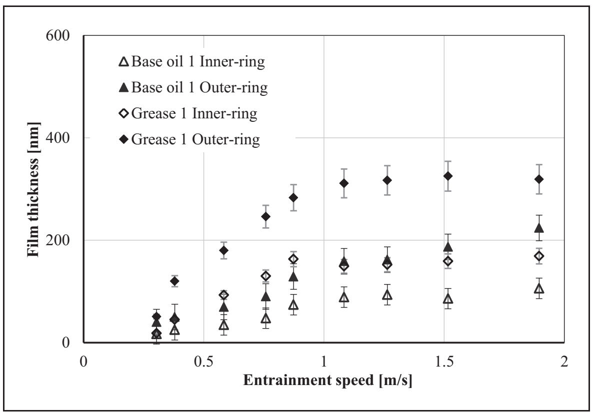

Figure 11. Base oil and film thickness of grease 1.

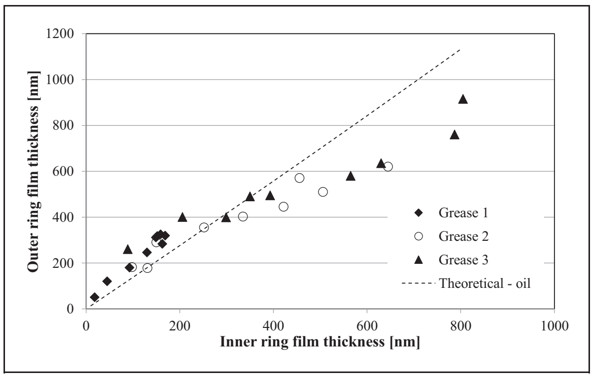

Figure 12. Outer ring film thickness vs inner ring film thickness for three greases.

Newtons. It is clear that in the case of these experiments the centrifugal effect can be neglected.

Conclusions

An original test rig, based on electrical capacitance method, was successfully employed to facilitate and extend the study into the measurement of lubricant film thickness in real rolling element bearing EHD contacts. With its novel capability of this rig to measure lubricant film thickness at inner ring and outer ring of the tested bearing, film thickness measurements of three types of greases with different constituents and their corresponding base oil were successfully conducted under variable rotational speeds.

For the part of base oils measurements, film thickness extracted from capacitance measurement is smaller than the theoretical one. The fact is at thin films, the method of extracting film thickness from capacitance is very sensitive to the exact dimensions of the contact and to the contribution to the total capacitance of the region outside of the contact. At lower speeds, the difference between outer and inner film thickness is on average about 38%. It was found that the ratio of the outer ring film thickness and inner ring film thickness follow a general trend obtained from the difference of the inlet geometries of the two contacts.

The film thickness formed by the greases was larger than that of the base oil for the entire range of the entrainment speeds employed in these experiments. It was also observed that the ratio between the outer ring contact film thickness and inner ring contact film thickness changes in a range larger than that found for the base oils. It is believed that this was due to a number of factors such as the use unworked grease, vibrations of the test rig, adherence of thickener to the track or passage of lumps of thickener through the loaded area.

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